

Structural Analysis of Ground Articulating Pipeline System

Ying Li^{*1}, Samuel Frimpong²

¹Houston, TX, USA; ²University of Missouri-Rolla, MO 65401, USA

^{*}liyinglzh@yahoo.com; ²frimpong@met.edu

Received 9 April 2014; Accepted 22 April 2014; Published 26 June 2014

© 2014 Science and Engineering Publishing Company

Abstract

The stress and deflection of the ground articulating pipeline (GAP) system are analyzed based on the theory of mechanics of materials. The governing equations of the pipe stress, deformation and rotation are established according to Hooke's law and double-integration method. The pipeline unit is modeled as a multi-physics three dimensional (3D) model with the load, temperature and pressure. ASME B31.1 is specified as the applicable piping code. The pipe system stress and deflection distributions are analyzed in two loading cases, which are (i) dead weight, internal pressure and thermal and (ii) dead weight, internal pressure, thermal and moment. The results show that pipeline unit has a smaller stress and deflection distribution in case 1 than case 2. The study provides a solid foundation for the optimization design of the GAP system.

Keywords

Ground Articulating Pipeline; Multi-physics Modeling; Stress; Deflection

Introduction

The GAP technology is an innovation design. It will be used to create and transport oil sand slurry from production faces through flexible pipeline system to the existing hydro-transport system. A lot of efforts have been made to conceptualization of the GAP mechanical system [Frimpong et al., 2002; Li et al., 2007]. Kinematics and dynamics simulations are being carried out to define and analyze system motion, force and torque for examining conceptual design and analyzing performance of the GAP system based on the multi-body dynamics [Li et al., 2007]. A virtual prototype simulator has been developed for simulation of the velocity and pressure distributions for all components of the GAP flow system [Frimpong et al., 2003]. Further investigation will be made on the strength examination of various GAP components, which are very important to determine the bearing

capacity of water and slurry pipeline and the productivity of the GAP system.

There are two major problems in examining component strength of the GAP pipeline system. One is the pipeline stress. Another one is the pipeline deflection including the deformation and rotation. The stress and deflection behavior in many cases yields to the failure of components or collapse of an entire system. The dead weight and load on the pipeline cause it to fail and deform. Slurry and water flow induce the pressure and temperature to the pipeline. They create additional stress in the piping system.

Multi-physics simulation includes such factors as stress and strain as well as thermal, fluid flow and pressure effects [Cross and Wheeler, 2000]. The mechanics of materials [Roark and Yong, 1975; Fairman and Cushall, 1953] are used to analyze the stress and deflection of the pipe. The commercial software Pipe Pack, which is a part of ALGOR software [Algor Uses Guide, 2006] offers the multi-physics simulation of, pipeline system. The software provides the structural analysis with various industrial standards piping codes.

The paper only focuses on developing a multi-physics model of the dual ball joint pipeline to simulate static stress and deflection. It considers the effect of the temperature and pressure caused by the water and slurry. The code that is applied is the ASME B31.1 power piping code [The American Society of Mechanical Engineers, 2001]. The following work has been carried out: (i) introduce the theory background of the stress and deflection of pipeline; (ii) create 3-D solid visualization multi-physics models; (iii) calculate the pipe system stress and deflection distribution; and (vi) determine the maximum stress, deformation and rotation.

Conceptual GAP System Description

The GAP system will facilitate the conveyance of oil sands slurry via articulated pipeline system to a fixed pipeline. FIG. 1 shows a conceptual design model of the GAP system [Li et al., 2007]. The GAP system consists of a series of rigid pipelines, including the water pipelines and slurry pipelines, one fixed pipeline and one mobile slurry system. The shovel excavates and feeds dry oil sands lumps into the mobile slurry system with the addition of hot water into the system, and the oil sands are slurried. The resulting oil sands slurry is then pumped through the GAP system to the fixed pipeline. A unique dual ball joint assembly is required to allow flow of both slurry and fresh water while permitting the position changes between adjacent pipelines.

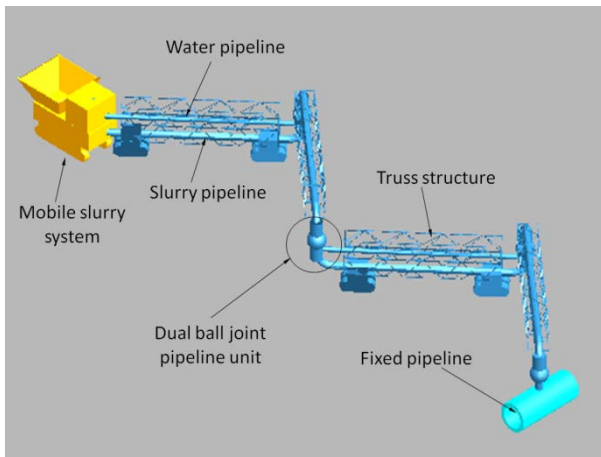


FIG. 1 CONCEPTUAL DESIGN MODEL OF THE GAP SYSTEM (LI ET AL., 2007)

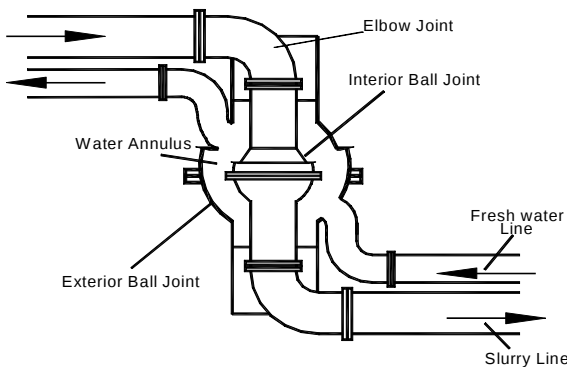


FIG. 2 DUAL BALL JOINT PIPELINE UNIT OF THE GAP SYSTEM (FRIMPONG ET AL., 2002)

FIG. 2 shows the geometric structure of the dual ball joint pipeline unit of the GAP system. Water lines and slurry lines are connected by elbow joints and dual ball joints. The dual ball joints conceptually contain an internal ball joint and co-axial external ball joint. The two balls form an annulus for conveyance of water. An internal ball joint supports the flow of slurry while

the external joint secures the flow of fresh water to be used in the slurification of oil sands. The dual ball joint assembly will support the articulation of the pipe arms to enable the adjustment of the working length.

Background Theory of the Pipe Stress and Deflection

The stress and deflection equations of the GAP pipeline system are established under the assumptions: (i) the pipeline system is subjected to static load, which is gradually applied; (ii) the temperature and pressure of water and slurry are constant through the pipe length; (iii) the water and slurry mixture flow is continuous from one end to other end; (iv) the proportional elastic limit of the pipe material is not exceeded and Hooke's law applies; (v) the modulus of elasticity is the same for both tension and compression; and (vi) the pipeline material is homogeneous.

General equations for axial stress, bending stress and torsional stress are calculated according to the formulae 1-3 [Fairman and Cushall, 1953], respectively.

$$S_a = \frac{F_a + P_a}{A_c} \quad (1)$$

$$S_b = \frac{\left[(i_{ip} M_{ip})^2 + (i_{op} M_{op})^2 \right]^{1/2}}{Z} \quad (2)$$

$$S_t = \frac{M_t}{2Z} \quad (3)$$

where F_a is the axial force, P_a is the axial force from internal pressure, A_c is the corroded cross-section area, i_{ip} is the in-plane stress intensification factor, M_{ip} is the in-plane moment, i_{op} is the out-of-plane stress intensification factor, M_{op} is the out-of-plane moment, M_t is the torsion, and Z is the section modulus or effective section modulus.

Pipe deformation j is calculated as general differential equation 4 according to double-integration method [Fairman and Cushall, 1953]. In applying this equation, coordinate axes must be chosen with the i axis parallel to the longitudinal axis of the pipe. M_i is the bending moment at a section whose distance from the origin is i , and j is the j axial deflection of the pipe at that section from the k axis. Two consecutive integrations of the differential equation give the equation of the pipe in terms of i and j .

$$\frac{d^2 j}{di^2} = \frac{M_i}{EI} \quad (4)$$

$$I = \pi \frac{(D^4 - d^4)}{64} \quad (5)$$

where E is the young's modulus, I is the moment of inertia, D is the outside diameter and d is the inside diameter.

Two transverse sections of the pipe at distance di apart will be included to each other at angle $d\theta$ after bending. Pipe rotation $d\theta$ is obtained as equation 6 by using area-moment method [Fairman and Cushall, 1953].

$$d\theta = \frac{1}{EI} \times M_i di \quad (6)$$

The rotation angle θ between the tangents to the pipe elastic curve at points A and B is obtained by integration of equation 7.

$$\theta = \frac{1}{EI} \int_A^B M_i di \quad (7)$$

GAP Multi-Physics Modeling

The dual ball joint pipeline unit is an important component and also is easy to failure in the GAP system. The modeling, simulation and analysis of the pipeline unit are given below.

TABLE 1 PIPE SPECIFICATIONS

	Slurry pipe	Water pipe
Inside fluid (Frimpong et al., 2002)	Oil sand slurry	Fresh water
Pipeline material (Mohinder, 2000)	Low C.S. A106B (Carbon content below .30%)	Low C.S. A106B (Carbon content below .30%)
Allowable stress code (Mohinder, 2000)	A106 B98	A106 B98
Angle range θ_3 (degree) (Li et al., 2007)	90	90
Outer diameter (m) (Frimpong et al., 2002)	0.62	0.45
Unit length (m) (Li et al., 2007)	15	15
Fluid density (kg/m ³) (Frimpong et al., 2002)	1.6	1.0
Pipe wall thickness (m) (Mohinder, 2000)	0.00952	0.00952
Insulation thickness (m) (Mohinder, 2000)	0.015	0.015
Insulation density (kg/m ³) (Mohinder, 2000)	35	35
Design pressure (kg/m ²) (Frimpong et al., 2003)	0.170	0.055
Design temperature (°C) (Frimpong et al., 2002)	70	70
Expansion (m/m) (Mohinder, 2000)	0.0012	0.0012

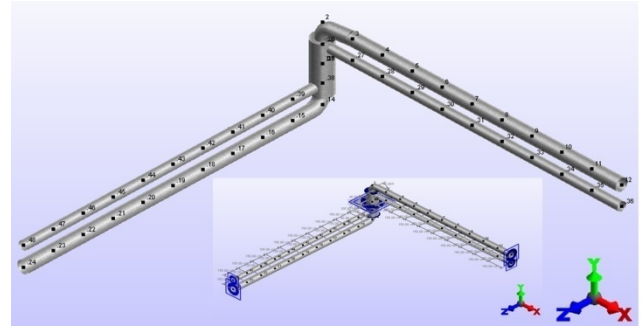


FIG. 3 PIPELINE MODEL WITH POINT LOCATION AND DIMENSIONAL SYMBOLS

Modeling of Dual Ball Joint Pipeline

FIG. 3 shows that a finite element multi-physics model of a 3D dual ball joint pipeline is created in the reference coordinate system $O(X,Y,Z)$. The model consists of two water straight segments and two slurry straight segments connected with a dual ball joint. There is 90 degree bend between two water or slurry pipelines. The coordinate system is attached to the model at any arbitrary point, the x-axis is the longitudinal axis of the right pipeline unit, the y-axis is the longitudinal axis of the left pipeline unit, and the positive z-axis is directed upwards. The water or slurry line unit is discretized into 10 segments displayed with the point location, dimensional symbols, anchors and flanges. Boundary conditions are such that the dual ball joint pipelines are rigidly constrained by points 12, 13, 25, 24, 36 and 48. The water pipelines are supported by the steel supports which are connected to the truss at points 36 and 48. The slurry pipelines are supported by the steel supports, which are connected to the truss at points 12 and 24. The dual ball joint are supported by the steel supports at points 13 and 25. The model is loaded by the weight of pipelines and other components. The effects of pressure and temperature on load have been considered.

TABLE 1 summaries the properties of the pipeline [Frimpong et al., 2002; Li, Y. et al., 2007; Frimpong et al., 2003; Mohinder, 2000]. The pipe material is selected according to standard Low C.S. A106B pipe, which can carry slurry and water at elevated pressure and temperature. The pipe also meets the American Society for Testing and Materials (ASTM) standard A106 B98, which may be used in temperatures of up to 538 °C, at various American Society of Mechanical Engineers (ASME) code stress levels [The American Society of Mechanical Engineers, 2001]. The slurry pipe dimensions are given as the diameter of 0.62 m, thickness of 0.00952 m and length 15 m. The water

pipe dimensions are designed as the diameter of 0.45 m, thickness of 0.00952 m and length 15 m. The piping system mass includes the mass of the steel piping, piping insulation and piping fluid. The insulation layer is the poly-urethane with a density of 35 kg/m³ and the thickness is 0.015 m. The fluid is the slurry and water with the densities of 1.6 and 1.0 kg/m³ at the internal pressure of 0.17 and 0.055 kg/m², respectively. The temperature and expansion of the fluid are 70 °C and 0.0012, respectively.

Analyzing of Dual Ball Joint Pipeline

The structure analysis is performed to determine the pipeline stresses, rotations, and deformations. According to load level, two cases will be discussed below.

In the first case, the loads applied to the model include the dead weight of the dual ball joint pipeline, internal pressure and thermal. Adapting the design parameters reported in TABLE 1, the distributions of the model stress, rotation and deformation are obtained in FIGs. 4-6.

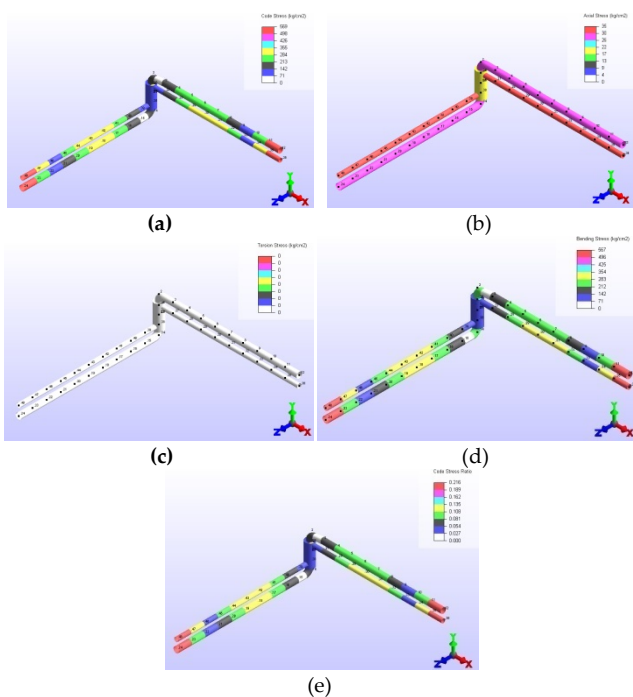


FIG. 4 PIPELINE STRESS DISTRIBUTIONS: (A) CODE STRESSES; (B) AXIAL STRESSES; (C) TORSION STRESSES; (D) BENDING STRESSES; AND (E) RATIOS BETWEEN CODE STRESS AND ALLOWABLE STRESS

FIG. 4 indicates the stress distributions of the dual ball joint pipeline. FIG. 4a shows the code stress profile. It undergoes a maximum value of 569 kg/cm² at segments 12, 24, 36 and 48 of the water and slurry lines and a minimum value of 71 kg/cm² at segments 3

and 15, which are near to the elbow joints of the slurry pipeline. FIG. 4b gives the axial stress contour. The maximum value of 35 kg/cm² locates at the water pipeline and the minimum value of 22 kg/cm² does at the dual ball joint. Fig. 4.7c illustrates torsion stress distribution, which keeps 0 in this case. Fig. 4d displays the bending stress profile. The maximum value of 567 kg/cm² occurs at segments 12, 24, 36 and 48 and the minimum value of 71 kg/cm² does at segments 3 and 15. Fig. 4e depicts the ratio contour between the code stress and allowable stress. The stress to allowance ratio is the division of the maximum allowance stress per ASME code B31.1 and the maximum actual stress per segment. The maximum ratio stress of 0.22 is found at segments 12, 24, 36 and 48 and the minimum one of 0.027 is done at segments 3 and 15. This result shows the maximum bending stress gives the greatest contribution to the maximum code stress. The maximum code stress is smaller than the allowable stress.

FIG. 5 indicates the displacement profile of the dual ball joint pipeline. FIG. 5a depicts that the displacement magnitudes decrease from the center to two ends of each pipeline unit. The maximum displacement magnitude of 1.072 cm appears at segments 31 or 43 and the minimum one of 0 cm appears at segments 12, 14, 24, 26, 36 and 48. FIG. 5b indicates the displacement distribution in x axis. The slurry pipeline displacement values decrease from segments 3 to 12 and the water line displacement values decrease from segments 27 to 36. The displacement keeps constant value at the left pipeline. The maximum displacement value of 0.025 cm appears at segment 13 while the minimum one of 0 cm appears at the left pipelines. FIG. 5c shows the y axial displacement profiles of the model. It can be noted that all displacement values are along with negative y axis. The displacement values of each pipeline unit decrease from the center to two ends. There is a maximum displacement value of -1.072 cm at segments 31 and 43 and the minimum displacement value of 0 cm at segments 12, 13, 24, 36, 38 and 48. FIG. 5d gives the displacement distribution in z axis. The displacement values increase from segments 34 or 48 to segments 15 or 39 at the left pipeline units, respectively while the displacement keeps constant value at the right pipeline units. The maximum displacement value of -0.025 cm appears at segment 39 and the minimum one of 0 cm appears at the right pipeline units.

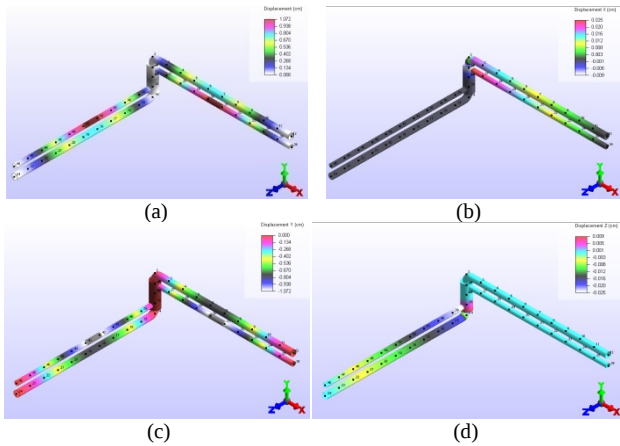


FIG. 5 DISPLACEMENT DISTRIBUTIONS: (A) DISPLACEMENT MAGNITUDE; (B) DISPLACEMENT IN X AXIS; (C) DISPLACEMENT IN Y AXIS; AND (D) DISPLACEMENT IN Z AXIS

FIG. 6 displays the rotation contours in x, y and z axis. Rotation direction is positive sign when clockwise and negative sign when counterclockwise. In FIG. 6a, the rotation distribution is presented in x axis. It can be seen that at the left water pipeline, the rotation values decrease from segments 39 to 45, which have positive clockwise followed by an increase from segments 45 to 48, which are counterclockwise. At the left slurry pipeline unit, the rotation values with positive clockwise decrease from segments 15 to 19. Then, the rotation values with negative counterclockwise increase from segments 19 to 22. Finally, the rotation values with negative counterclockwise decrease from segments 22 to 24. It can also be seen that the rotation keeps a constant value at the right pipeline units and dual ball joint. The maximum rotation value of 0.137 degree appears at segment 39 and the minimum one of 0 degree appears at segments 19 and 43. In FIG. 6b, the rotation distribution is given in y axis. It is noted that the rotation value doesn't exist. In FIG. 5c, the rotation contour is obtained in z axis. It can be seen that the rotation values increase from segments 2 to 9 then decrease from segments 9 to 12 at the right slurry pipeline. At the right water pipeline, the rotation values have negative counterclockwise and decrease from segments 27 to 31. Then, they are positive clockwise and increase from segments 31 to 34. Finally, they are also positive clockwise and decrease from segments 34 to 36. The rotation keeps constant value at the left pipelines and dual ball joint. The maximum rotation value of 0.112 degree appears at segment 34 and the minimum one of 0 degree appears at segments 6 and 30.

In the second case, the loads applied to the model include the dead weight of the dual ball joint pipeline,

water and slurry pressure, thermal and moment. The moment of $9E+004$ Nm is applied to the dual ball joint pipeline at segments 15 and 39 (see FIG. 3) in y axial direction, respectively. The design parameters are given in TABLE 1. The pipeline stresses (FIG. 7), rotations (FIG. 8), and deformations (FIG. 9) are analyzed below.

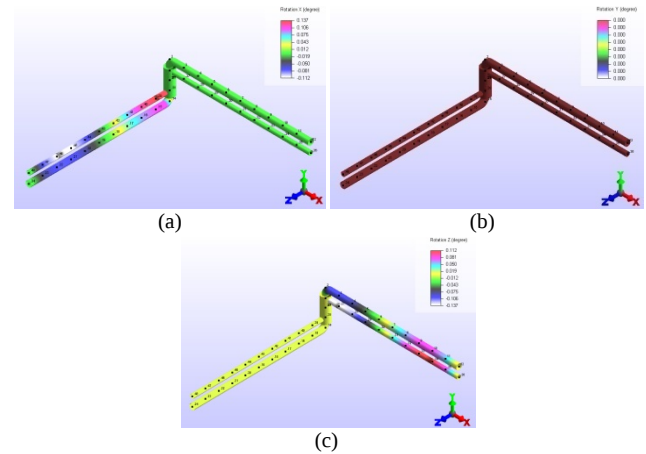


FIG. 6. ROTATION DISTRIBUTIONS ALONG THE PIPING SYSTEM: (A) ROTATION IN X AXIS; (B) ROTATION IN Y AXIS; AND (C) ROTATION IN Z AXIS

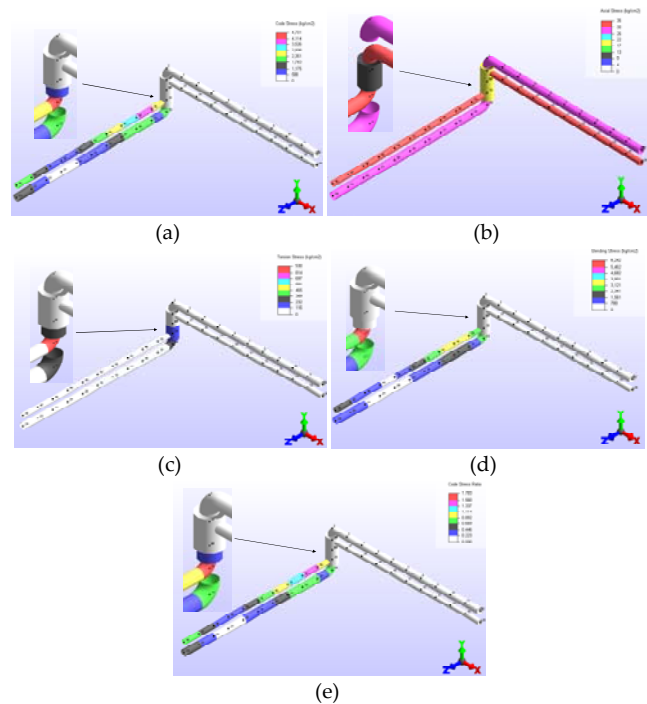


FIG. 7 PIPELINE STRESS DISTRIBUTIONS: (A) CODE STRESS; (B) AXIAL STRESS; (C) TORSION STRESS; (D) BENDING STRESS; AND (E) RATIO BETWEEN CODE STRESS AND ALLOWABLE STRESS

FIGs. 7a-e display the distributions of the code stress, axial stress, torsion stress, bending stress and ratio, respectively. FIG. 7a describes the code stress profile. The code stress undergoes a maximum value of 4701 kg/cm^2 at the elbow joint 38 and a minimum value of 588 kg/cm^2 at segments 21, 22, external joint 1 and

right pipelines. FIG. 7b depicts the axial stress distribution. It exhibits apparent symmetry distribution. There is a maximum axial stress value of 35 kg/cm^2 at the water pipeline and a minimum value of 13 kg/cm^2 at the internal ball joint. FIG. 7c illustrates the torsion stress distribution. The maximum torsion stress value of 930 kg/cm^2 occurs at segment 38 and the minimum one of 116 kg/cm^2 does at the ball joint 1 and straight pipelines. Fig. 7d displays the bending stress distribution. The maximum bending stress value of 6242 kg/cm^2 appears at the elbow joint 38 and the minimum one of 780 kg/cm^2 does at segments 20, 21, 22, 44, dual ball joint and right pipeline units. FIG. 7e gives the ratio contour between the code stress and allowable stress. The maximum ratio stress of 1.783 is obtained at the elbow joint 38 of the water pipeline and the minimum one of 0.223 is done at the segments 21, 22, external joint 1 and right pipeline units. It can be noted that the bending stress gives the greatest effect on the code stress. The maximum code stress is greater than allowable stress.

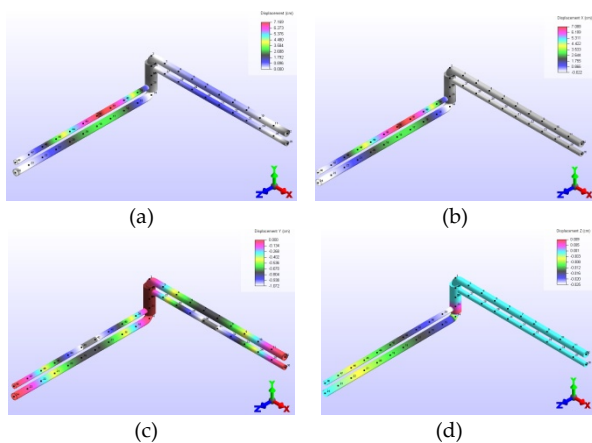


FIG. 8 DISPLACEMENT DISTRIBUTIONS: (A) DISPLACEMENT MAGNITUDE; (B) DISPLACEMENT IN X AXIS; (C) DISPLACEMENT IN Y AXIS; AND (D) DISPLACEMENT IN Z AXIS

FIGs. 8a-d indicate the displacement magnitude, x axial displacement, y axial displacement and z axial displacement of the dual ball joint pipeline, respectively. FIG. 8a depicts the pipeline displacement magnitude profile. It can be seen that the displacement magnitudes decrease from the pipe centers to two ends for each pipeline unit. The maximum displacement magnitude of 7.169 cm appears at segment 42 and the minimum one of 0 cm appears at the two ends. FIG. 8b indicates the displacement distribution in x axis. The displacement values decrease from the centers to the pipe ends at the left pipeline units and the displacement almost keeps a constant value at the right pipelines and dual ball joint.

There are the maximum displacement value of 7.088 cm at segment 42 and the minimum one of 0 cm at the pipes 14, 24, 38 and 44. FIG. 8c shows the displacement contour in y axis. It can be seen that the displacement values present negative in y axial direction and decrease from the center to two ends for each pipeline unit. There are a maximum displacement value of -1.072 cm at segments 31 and 43, and a minimum displacement value of 0 cm at segments 2, 12, 14, 24, 36, and 48. FIG. 8d gives z axial displacement distribution of the dual ball joint pipeline. The displacement values increase from segments 24 or 46 to 15 or 39 at the left pipeline units, respectively and the displacement keep constant value at the right pipeline units. The maximum displacement value of -0.025 cm occurs at segment 38 and the minimum one of 0 cm appears at the right pipeline units.

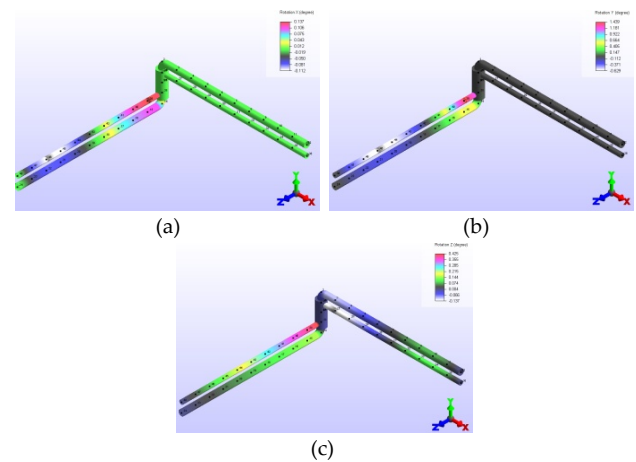


FIG. 9 ROTATION DISTRIBUTIONS: (A) ROTATION IN X AXIS; (B) ROTATION IN Y AXIS; AND (C) ROTATION IN Z AXIS

FIGs. 9a-c display the rotation contours in x, y and z axis, respectively. It should be noted that a positive rotation indicates clockwise while a negative one does counterclockwise. In FIG. 9a, the rotation distribution is presented in x axis. It can be seen that at the left water pipeline, the rotation values are negative counterclockwise and increase from segments 48 to 45. Then, positive clockwise rotations present and increase from segments 45 to 39. At the left slurry pipeline, the rotation values with the negative counterclockwise increase from segments 24 to 20 and the rotation values with the positive clockwise increase from segments 20 to 15. The rotation almost is a constant value at the right pipeline units and dual ball joint. The maximum rotation value of 0.137 degree appears at segment 39 and minimum one of 0 degree appears at segments 24, 18, 43, 29 and the right pipeline unit. In FIG. 9b, the rotation distribution is given in y axis. At the left water line, the rotation

values are negative sign from 18 to 42, which increase from segments 48 to 44 and decrease from segments 44 to 42. Then, the rotation values change into positive sign and increase from segments 42 to 39. At the left slurry line, the rotation values present negative counterclockwise from segments 24 to 18, which increase from segments 24 to 21 and decrease from segments 21 to 18. Then, the rotation values present positive clockwise and increase from segments 18 to 15. There is the maximum rotation value of 1.439 degree at segment 39 while the minimum one of 0 degree at the right pipeline units. In FIG. 9c, rotation contour is obtained in z axis. It can be seen that the rotation values are positive clockwise and increase from segments 48 or 24 to 39 or 14 at the left pipelines. The rotation values almost keep a constant at the dual ball joint. It can also be seen that the rotation values decrease from segments 2 to 10 and increase from segments 10 to 12 at the right slurry line. The rotation values decrease from segments 27 to 33 then increase from segments 33 to 36 at the right slurry line. The maximum rotation value of 0.425 degree appears at segment 39 and the minimum one of 0 degree appears at segment 31.

TABLE 2 summarizes the maximum stress in two cases. The results show that maximum axial stress is the same for both case 1 and case 2. Then, the maximum bending stress is smaller in case 1 than in case 2, which occur at segments 36 and 38, respectively. The maximum torsion stress is higher in case 2 than in case 1, which experiences by the zero in case 1 and with the 938 kg/cm² in case 2. The structure experiences greater bending and torsional stress changes due to the moment applied, but the changes in axial stress are identical. There is a stress peak point at segment 38. This means that there are some factors increasing the stress in this location, which are the force, flange and anchor. By comparing FIGs. 4a with 7a, it is found that segment 38 at case 2 creates an increase of stress due to the moment to be applied. The segment 38 reduces stress in case 1 because of the zero moment.

TABLE 3 summarizes the maximum displacement and rotation obtained in two cases. By comparing the maximum displacement in case 1 with case 2, it can be noted that the displacement magnitude in case 2 is much greater than in case 1 owing to moment involving. Since the moment is applied to the dual ball joint pipeline at segments 15 and 39 in the y axial direction, the x axial displacement in case 2 is greater than in case 1. However, the y or z axial displacements

in two cases are identical. In the x axial direction, the maximum rotation in case 1 is the same compared to case 2. But, in the y axial direction, the maximum rotation in case 2 is greater than case 1 due to the moment applied to the model at segments 15 and 39 in the y axial direction. The maximum rotation presents the zero in case 1 and with the 1.439 degree in case 2. In the z axial direction, the maximum rotation value of -0.137 degree in case 1 is counterclockwise then the maximum rotation value of 0.425 degree in case 2 is clockwise. The structure experiences greater displacement changes due to the moment applied, but the changes in rotation are smaller. There is a displacement peak point at segment 42. This means that there are some factors increasing the displacement in this location, which are the force and anchor. By comparing FIGs. 5a with 9a, it is found that segment 42 at case 2 creates an increase of displacement due to the moment applied to segment 39. The segment 42 reduces displacement in case 1 because of the zero moment. The maximum displacement magnitude in case 1 is only 1.072cm in segment 31. The maximum displacement in case 2 is 7.169 cm in segment 42. This number is significant to identify the weakest elements.

TABLE 2 COMPARISON OF STRESS RESULTS FOR TWO CASES

Maximum stress (kg/cm ²)	Case 1		Case 2	
	Axial stress	35	Segment 26	35
	Bending stress	567	Segment 36	6242
	Torsional stress	0		930
	Code stress	569	Segment 36	4701
	Stress ratio	0.22	Segment 36	1.78

TABLE 3 COMPARISON OF THE DEFLECTION RESULTS FOR TWO CASES

Max Dis. (cm)	Mag-nitude	1.072	Segment 31	7.169	Segment 42
	x	0.025	Segment 26	7.088	Segment 42
	y	-1.072	Segment 31	-1.072	Segment 31
	z	-0.025	Segment 38	-0.025	Segment 38
Max Rotation (degree)	x	0.137	Segment 39	0.137	Segment 39
	y	0		1.439	Segment 39
	z	-0.137	Segment 27	0.425	Segment 38

Conclusions

The stress and deflection of the dual ball joint pipeline is studied by developing a multi-physics simulation model based on the theory of mechanics of materials, FEA and some assumptions. The dual ball joint pipeline unit is modeled in Algoreenvironment with the boundary conditions. The model integrates finite element mesh and multi-physics factors with the load, temperature and pressure. ASME B31.1 is specified as the applicable piping code. By the simulation and

animation of model, the stress and deflection distributions are presented in two load cases. The stress and deflection are analyzed in 3D space. In the first case, the loads applied to the model include the dead weight, internal pressure and thermal. The maximum code stress is 569 kg/cm² with the stress ratio of 0.22. The maximum displacement magnitude is 1.072 cm. In the second case, the loads applied to the model include the dead weight, internal pressure, thermal and moment. The maximum code stress is 4701 kg/cm² with the stress ratio of 1.78. The maximum displacement is 7.169cm. By comparing the stress and displacement in case 1 with case 2, it can be noted that for the static stress case the moment has a greater effect on the pipeline strength, increasing the stress and displacement in almost all the segments.

The further research will involve dynamic analysis of the GAP piping system to optimize the structural design of the GAP system.

REFERENCES

- Algor User's Guides. "Stress Calculation." Technical Manuals, 2006.
- Cross, M. and Wheeler, D. "The Role of Multi-physics simulation in virtual manufacturing, Virtual Design and Manufacture." ImechE Seminar Publication, Vol. 18, 2000, pp. 11-27.
- Fairman, S. and Cushall, C.S. "Mechanics of Materials." John Wiley & Sons, Inc. New York, 1953.
- Frimpong, S., Changirwa, R.M.M., Asa, E. and Szymanski, J. "Mechanics of Oil Sands Slurry Flow in a Flexible Pipeline System." IJSM.16, 2002, pp. 105-121.
- Frimpong, S., Oluropo, R.A. and Szymanski, J. "NUMSOSS: Numerical simulation software for oil sand slurry flow in flexible pipelines." SCSC 2003 of the Society for Modeling and Simulation International, Montreal, Quebec, Canada, 2003, pp.145-154.
- Li, Y., Szymanski, J. and Frimpong, S. "Preliminary Simulation of the GAP Mechanical System for Oil Sands Haulage." International Journal of Mining, Reclamation and Environment, Vol. 21, No. 4, 2007, pp. 295 – 305.
- Mohinder, L.N. "Piping Handbook 7th ed." McGraw Hill, New York, 2000.
- Roark, R.J. and Yong, W.C. "Formulas for Stress and Strain, 5th ed." McGraw-hill, New York, 1975.
- The American Society of Mechanical Engineers. "ASME B31.1 – 2001 Edition Power Piping." ASME, New York, 2001.
- Ying Li** has focused on mining machinery research since 1997 as a Senior Staff Engineer at the Caterpillar Global Mining, USA; a Post-doctor Research Fellow at the University of Missouri-Rolls, USA; a Post-doctor Research Fellow at the University of Alberta, Canada; and a PhD Candidate at the University of Science and Technology Beijing, China. Her research interests include mining equipment dynamics modeling, virtual prototype simulation and structural strength analysis. She has published more than 60 papers on mechanical and mining science and engineering in journals and conference proceedings.
- Samuel Frimpong** is a Professor of Mining Engineering and the Robert H. Quenon Endowed Chair at the Missouri University of Science and Technology and an Adjunct Professor at the University of Alberta. He holds PhD from the University of Alberta. His current research areas include surface mining, machine-formation interactions, mining machine health and longevity, oil sands extraction, bulk materials handling, advanced stochastic modeling and risks and safety engineering. He is a Registered Professional Engineer with memberships in APEGGA, CIM, SME, ASCE, and SCS.